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Effects of paddle tip-clearance on performance of a forage blower

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EFFECTS OF PADDLE TIP-CLEARANCE ON
PERFORMANCE OF A FORAGE BLOWER

by

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INTRODUCTION

The Forage Blower

The machine which is commonly known as a forage blower is not a blower in the usual sense of the word. Whereas a blower for pneumatic conveying depends strictly on the suspension of particles in a high velocity air stream, the forage blower's conveying action depends to a greater extent on the kinetic energy imparted to the forage by contact with the blower paddles. Segler (17, p.661) stated that,

> the working of the silo [forage] blower is based on principles very different from those of a fan designed for pneumatic conveying, so that the known and published data for calculations in connection with fans is [sic] not applicable to this kind of conveyor.

Thus, the forage blower might have been aptly called a "forage thrower".

The forage blower is by far the most common device for elevating chopped forages into upright silos and is also used extensively for placing both long and chopped hay into barn storage. There are no available figures as to the number of forage blowers on farms, for blowers were not considered in either the 1950 or 1954 census of agriculture. Some idea of the number of blowers may be obtained from the number of upright silos. The latest census figures from 1950 (18) show that there were in the United States 508,341 farms owning 614,027 upright silos. The National Association of Silo

Manufacturers (12) estimates that since 1950 there has been an average of 15,000 upright silos erected per year. Although there were doubtless some silos abandoned or salvaged during this period, there has probably been a sizeable increase in the total number. It seems reasonably safe to assume that most of the farms with an upright silo also have a forage blower.

In 1928, Duffee (9) described a new method which had been worked out at the University of Wisconsin for the handling of hay. The method consisted of field-chopping from the windrow and then blowing the chopped hay into the mow. This development was the advent of the modern field forage harvester which has almost entirely replaced the stationary ensilage cutter. It was, likewise, the beginning of the forage blower as we know it today.

The elevating of chopped material from the stationary ensilage cutter was done by paddles mounted on the periphery of the cutterhead. The present-day blower is essentially the old ensilage cutter with the cutting mechanism removed and with a different feed mechanism. There have, of course, been design changes and some improvement of performance over the years.

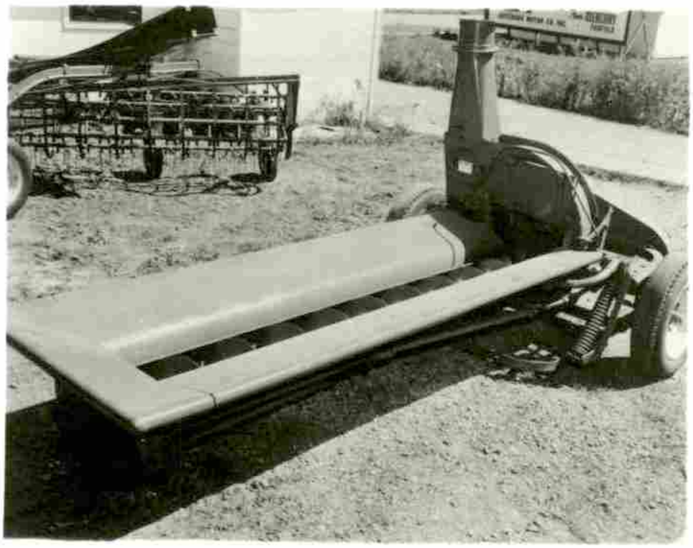
The forage blower is a simple and relatively low-cost machine. It consists generally of the following components: (a) an impeller assembly made up of a flywheel on which paddles are mounted; (b) a housing which fits fairly closely and usually concentrically around the impeller assembly and has a feed opening and a peripheral outlet; (c) a mobile frame in which the housing is mounted; (d) a feed mechanism which is usually hinged to the frame; and (e) a delivery pipe. Figure 1 shows a typical forage blower.

The simplicity due to few moving parts, the low cost, and a rather high capacity account for the popularity of the forage blower as compared to flight-type elevators for forage crops. A flight elevator of the type presently available, with a capacity in the range of a forage blower's capacity, would necessarily be large, cumbersome, and considerably more expensive than the blower.

The power-requirement and efficiency of the blower, however, are not so favorable. DeForest (4), in 1947, compared the performance of forage blowers with an experimental, high-speed, flight-type elevator. While elevating corn silage, the blower energy-requirements ranged from 0.269 to 1.598 horsepower-hours per ton. The energy-requirements for the experimental elevator, handling both corn and grass silage, ranged from 0.021 to 0.326 horsepower-hours per ton. Most present-day blowers require at least a two-plow tractor, or its equivalent, for a power source. Such a requirement practically rules out the use of the electric motor as a power source.

For finely-chopped forages, the general operating performance of blowers is quite satisfactory. However, the perform-

Figure 1. A typical forage blower



ance in handling long hay leaves much to be desired. Wedging of material within the housing and pipe blockages often occur. These longer materials are becoming more common with new trends in forage chopping and handling.

The forage blower is an important machine to a large group of farmers today. There is a definite need for improved performance with regard to both efficiency and to operational characteristics. Genuine improvements can be made only in the light of knowledge of the effects of pertinent variables on the forage blower performance.

Factors Affecting Blower Performance

There are a great many variable factors which can affect the performance of a forage blower to some degree. Following is a list of what are considered the pertinent variables:

1. housing diameter
2. housing shape
3. housing width
4. feed-opening size
5. feed-opening location
6. outlet size
7. outlet location
8. paddle size
9. paddle shape
10. paddle tip-clearance
11. paddle side-clearance
12. number of paddles
13. pipe diameter
14. pipe height
15. air pressure
16. blower speed
17. feed rate
18. crop type
19. crop moisture
20. crop length of cut

Relationships among and effects of some of the above variables are well known. Most have been considered to some extent in previous investigations, but there is very little reliable data to establish the effects of several of the variables on blower performance. Blower design has been based mainly on experimental information obtained from tests on stationary ensilage cutters and on "cut and try" procedures.

One of the variables which is known to be a very important factor in blower performance is paddle tip-clearance. Tip-clearance is the distance, generally fixed, between the outer end of the paddles and the rim of the blower housing. The usual design condition has been to have the tip-clearance as small as could be reasonably maintained with the type of construction used. There is little evidence to indicate that this close tip-clearance is necessarily the best. There is no available data on the effect of varying the tip-clearance over a reasonable range. It appears that more information on this important variable is needed for an enlightened approach to improving forage blower performance.

Review of Literature

A thorough search of agricultural engineering literature has revealed very few published reports of research on forage blowers, and none related to paddle tip-clearance, in the last 30 years. Thus, most of the research concerned with the blowing of forage into storage has been conducted on the stationary ensilage cutter.

In 1924 and 1925, the Agricultural Engineering Department at the University of Wisconsin conducted three series of rather extensive tests on stationary ensilage cutters. Duffee (6, 7 and 8) reported the results of these tests. Quite a few of the tests are applicable only to cutters, but several were concerned with elevation of the chopped material and are, hence, applicable to forage blowers.

The first series of tests by Duffee (6) was concerned with determining the efficiency of the ensilage cutters. He found that the power-requirement increased gradually with elevation. He determined the separate power-requirement for cutting and for blowing. On the cylinder-type machine with a separate blower, he used one electric motor for the cutter and another for the blower. On the flywheel-type machine, in order to determine the power required for blowing, he removed the knives and delivered chopped material from another machine. He found that the power required for blowing ranged from 36 to 72 per cent of the total requirement.

Duffee's second series of tests (7) was a study of some factors in the design of the cutting unit and from these tests, he developed a formula for the capacity of the cutters. The effects of pipe diameter, length of cut, speed, and feed rate were also studied. He also measured static pressure in the blower pipe and found that the most efficient elevating machine had the lowest static pressure. He concluded that this was proof of the lack of importance of air pressure in the conveying action of the ensilage cutter.

In the final series of tests, Duffee (8) considered the elevating function of the ensilage cutter in more detail, in addition to studying the cutting and feed mechanisms and standardization of size-rating and pipe connections. He found no significant difference in power-requirement for a four-paddle and a six-paddle blower on a cylinder-type cutter. As a result of observation and air-flow measurement in the discharge pipe, he proposed relocation of the discharge outlet.

In this series, which was performed with finely cut corn silage, he also gave considerable attention to the paddle clearance, although his data are rather limited. Two sets of paddles were used, one with the smallest possible tip-clearance and the other with from 1/8 inch to 1/4 inch tip-clearance. Duffee states that the performance of the close-fitting paddles was far superior to that of the other set. However, the difference in performance cannot be entirely attributed to tip-clearance, as there was also much less side-clearance on the close-fitting paddles. In another test, he had the housing close enough to the paddle tips, from the bottom around to the point of discharge, to carry out a 1/16 inch-thick washer. Duffee compared the performance at this clearance to a clearance of 1/8 inch at the discharge side and about 5/16 to 3/8 inch at the bottom. For these tests, he states that the small differences in results are due largely to speed variation.

Longhouse (10) studied the functional requirements of long hay blowers in 1949. The blower with which he worked was of the pneumatic type. Thus, data from his tests would not be generally applicable to the conventional forage blower.

Segler (17) in 1951 stated some rather basic facts on the functioning and design of ensilage cutters and blowers. He has been quoted earlier in this thesis on the principle of operation of the forage blower. He stated that air in the blower pipe actually retards the forage in the early stages of elevation, but, since the air-velocity is constant, does have some conveying action near the end of the pipe. Segler's only mention of paddle tip-clearance is a reference to Duffee's work which has been described above.

Objectives

This study was undertaken because of the need for improved performance of the forage blower. Paddle tip-clearance is one of the factors affecting blower performance about which there was little available data, but which was known to be one of the most important. It was felt that more information on paddle tip-clearance was necessary for a proper approach to the problem of improving blower performance.

The objectives of this study were:

1. To discover the relative importance of the clearance between the paddle tips and the blower housing in the satisfactory over-all operation of the forage blower.

2. To determine the effect of paddle tip-clearance on the fan characteristics of the blower.

3. To determine the effect of tip-clearance on the power-requirement, capacity, and efficiency of the blower for elevating chopped hay.

INVESTIGATION

Equipment

Preparation of the test blower

A John Deere Model 50 blower was made available by the John Deere Ottumwa Works for use in the investigation. This blower has four straight paddles, 6-5/16 inches wide at their outer ends, mounted on a square flywheel making up an impeller assembly which is approximately 34 inches in diameter. The housing is constructed with a slightly decreasing tip-clearance in the lower half and has a width of 7-1/16 inches. The impeller assembly is mounted on a 1-1/2-inch diameter shaft turning in two tapered roller bearings and driven by a flat belt.

Preparation of the test blower, other than instrumentation, consisted mainly of modification to allow for adjustment of the paddle tip-clearance and of construction of a circular housing concentric to the impeller shaft. These problems will be dealt with in the following paragraphs.

Tip-clearance adjustment. There appeared to be two feasible methods of changing the tip-clearance, both involving movement of the paddles. One method would be to have sets of paddles of varying lengths and to put a different set of paddles on the flywheel for each tip-clearance. The other method would be to move the paddles radially on the flywheel to obtain different clearances. The second method was considered more desirable if the paddle position could be made adjustable over a reasonable range of clearances. It was felt that an adjustment range from 0 to 1/2 inch would be sufficient.

Each paddle arm was bolted to the flywheel by four cap-screws. There was sufficient space between the bolt-holes in the paddle-arm to permit the machining of a slot to allow the necessary adjustment. Each hole in the paddle-arms was milled to form a slot as shown in Figure 2.

The slots provided for the mechanics of the adjustment. but a means of measuring or gaging the tip-clearanse had to be devised. An indirect method of measurement with some type of scale on the flywheel could have been worked out. However, it was felt that a direct gaging of the clearance would be less likely to permit error in measurement or in the making of an adjustment mechanism.

A set of 12 metal gage blocks were machined to the following thicknesses in inches: 1/32, 1/16, 3/32, 1/8, 5/32, 3/16, 7/32, 1/4, 5/16, 3/8, 7/16, and 1/2. For setting the tip-clearance, the gage block was placed in the bottom of the blower housing. The paddle was dropped down on the block and tightened in that position. All paddles were set in the same position, thus insuring the same clearance for all. The gage blocks are shown in Figure 3.

<u>Blower housing construction</u>. The rim for the new blower

Figure 2. Paddle arm with slots to permit adjustment of tip-clearance

Figure 3. Metal blocks used to gage paddle tip-clearance

housing was rolled in two parts with a radius of 17 inches. This forming was done by the John Deere Ottumwa Works with the maximum precision available with the rolling machine used to make the production blower rims. It was decided to make the housing front and back panels out of 1/2-inch thick plastic sheets. The material is commonly known as "Plexiglass." The visibility afforded by this transparent plastic was desirable to aid in making adjustments and in observing the operation of the blower.

The job of fitting the two parts of the rim in an accurate circle to the plastic sides and then to the blower frame in proper relation to the shaft center proved to be a difficult one. A plywood form was used to position the rim. After assembly of the housing and positioning of the impeller assembly by the use of shims under the bearing housing, a dial gage was used to check the accuracy of the housing rim.

With the bottom of the blower as a zero-reference, maximum plus and minus readings and other zero readings were taken and located around the rim. This was done 1 inch from the front, 1 inch from the back, and in the center of the rim. Maximum plus and minus readings were often only a short distance apart, pointing out the imperfection of the rolling process. There were also slight differences in radius at some points along a line from front to rear.

The 17-inch radius was maintained within reasonably close tolerances in the lower half of the housing, which is considered the most important part. The maximum plus reading was 0.018 inches and the maximum minus reading was 0.023 inches. Apparently due to imperfection in the fitting, the readings ran out to as much as + 0.064 at one point near the top of the housing. The position and magnitude of the gage readings are shown on the schematic drawings of Appendix A.

The clearance at the outlet cutoff point was thought to be a critical one. It was considered desirable to make possible the adjustment of this clearance independently of the clearance in the rest of the housing. For this reason, a 5/32-inch plate was installed in the housing outlet, to be raised or lowered in slots. This modification provided a variable cutoff point. With this plate in position, the outlet opening was 7-1/16 by 10 inches.

The completed blower is shown in a front view in Figure 4. The feed opening, which was used as shown, was 17-1/4 by 14 inches. The center of the housing was located 6-3/16 inches below the top of the opening and 6-1/4 inches from the left side of the opening.

Other equipment

The feed mechanism for the test blower was a raddle-type conveyor, 16 inches wide, 12 inches deep, and 21 feet long. The conveyor was driven by a tractor power-take-off and the speed was readily variable by adjustment of the tractor throttle. End gates were installed 20 feet apart in the conveyor to allow for uniform placement of forage over this length.

The blower discharge pipe was 8 inches in diameter. A 180-degree deflector was attached to the top section of pipe. Down-spouts were used to direct the forage into an enclosure. A utility pole was used to support the blower pipe and down-spouts.

A Cockshutt Model 40 tractor with a 6-cylinder diesel engine was used as a power unit for the blower. The drive was through a flat belt. The feed conveyor was driven from the power-take-off of an Oliver Model Super 55 tractor with a 4-cylinder gasoline engine.

Instrumentation

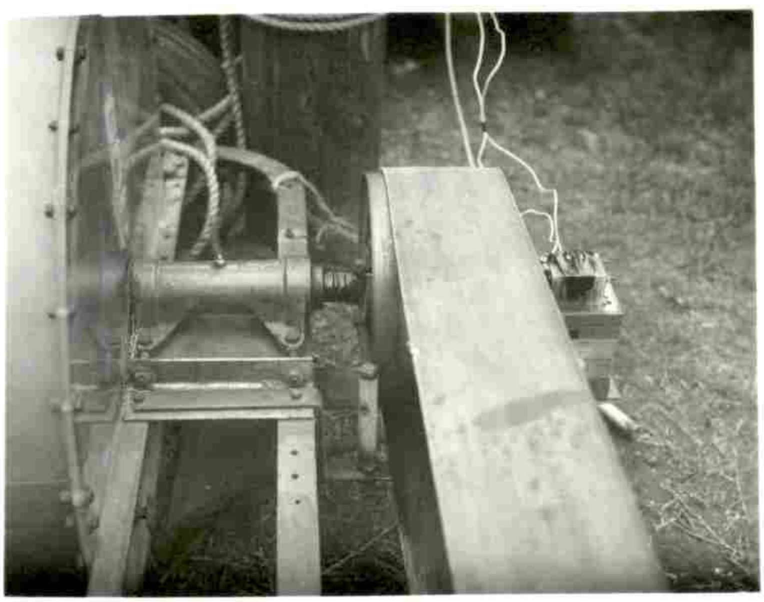
Instrumentation for power measurement

SR-4 electrical resistance strain gages were chosen as an indicator for the measurement of torque on the blower shaft. The use of the SR-4 gage for the measurement of torsional strain is a well known technique. The theory of this application is developed in several references (2, 5 and 13). The particular installation which was used is described below.

The strain gages were Baldwin Type C-7, with resistance of 500 $\pm$ 3 ohms, and gage factor of 3.18 $\pm$ 2 per cent. Four gages were bonded to the blower shaft along lines of principal strain, at equal intervals around the circumference. This installation, shown in Figure 5, was between the belt pulley

Figure 4. Front view of completed test blower

Figure 5. Strain gage installation between pulley and bearing-housing



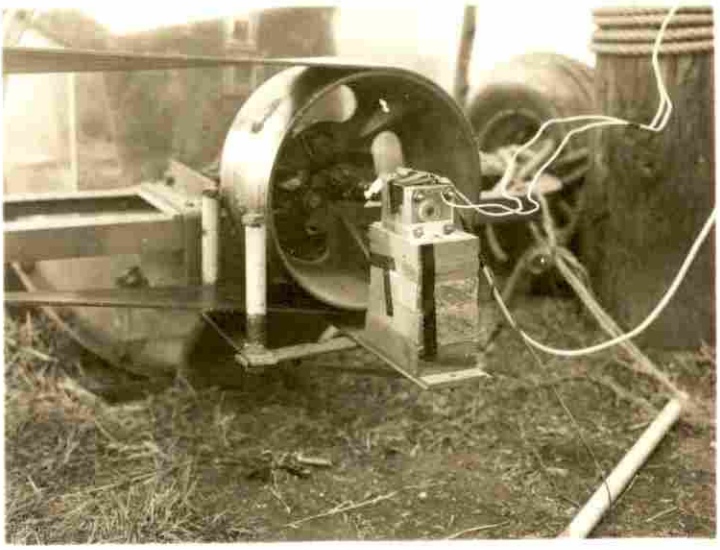
and the bearing housing. Using four active gages gave maximum strain amplification and compensated for both temperature and bending effects.

The electrical circuit through the gages on the rotating shaft was completed through a mercury-bath collector unit of the type used by Bockhop (2) and Burrough (3). The leads from the gages were brought into the collector unit through a hollow rotor shaft made of "Bakelite" tubing. The shaft size was 1/2-inch outside diameter and 1/4-inch inside diameter, based on Bockhop's experience of breakage of smaller plastic shafts (2, p.21). Each lead was connected to a copper disc mounted on the shaft. Each disc rotated in a separate cell, lined with a copper band and partly filled with mercury. The discs and bands were amalgamated with mercury prior to installation. The bands were attached to the body of the cell by small bolts which extended to the outside and served as external terminals. The body of the collector unit was constructed from 2-1/2 by 2-1/2-inch blocks of 1/4-inch "Plexiglass."

The collector unit rotor was driven by a flexible connection to the end of the blower shaft. A cam and breaker-points were mounted on the collector unit to actuate an event marker on the recording instrument. The collector unit installation is shown in Figure 6.

From the collector unit, the leads went to the recording instruments. These instruments consisted of a Brush Model BL 320 analyzer and a Brush Model BL 222 recording oscillograph.

Figure 6. Mercury-bath collector unit installed



The oscillograph used an electric stylus and BL 939 chart. The leads connected to the analyzer to complete a Wheatstone Bridge circuit. Unbalance of the bridge due to torque on the shaft was recorded by the oscillograph. This unbalance had been calibrated in pound-feet of torque. A sketch of the strain gage and instrument arrangement is shown in Figure 7.

A small event marker stylus on the recorder was connected in series with a 6-volt battery and with the breaker points on the collector unit. For each shaft revolution the circuit was broken once, causing a deflection of the event marker stylus. With the recorder chart moving at a known speed, the revolution marks were used to calculate shaft revolutions per unit time. The chart thus furnished a measure of torque and speed, the necessary information for calculation of power. See Appendix C.

Calibration of the strain gage installation was done by placing known weights on a torque arm attached to the blower shaft. Calibration was done prior to the tests and again following the tests. Results are shown in Figure 8. The slopes of the curves were used to convert lines deflection to torque. The calibration data are found in Appendix B.

Instrumentation for air-flow measurement

The quantities to be measured in the air-flow tests, in addition to power input and speed, were static pressure in the blower pipe and quantity of air delivered per unit time. A

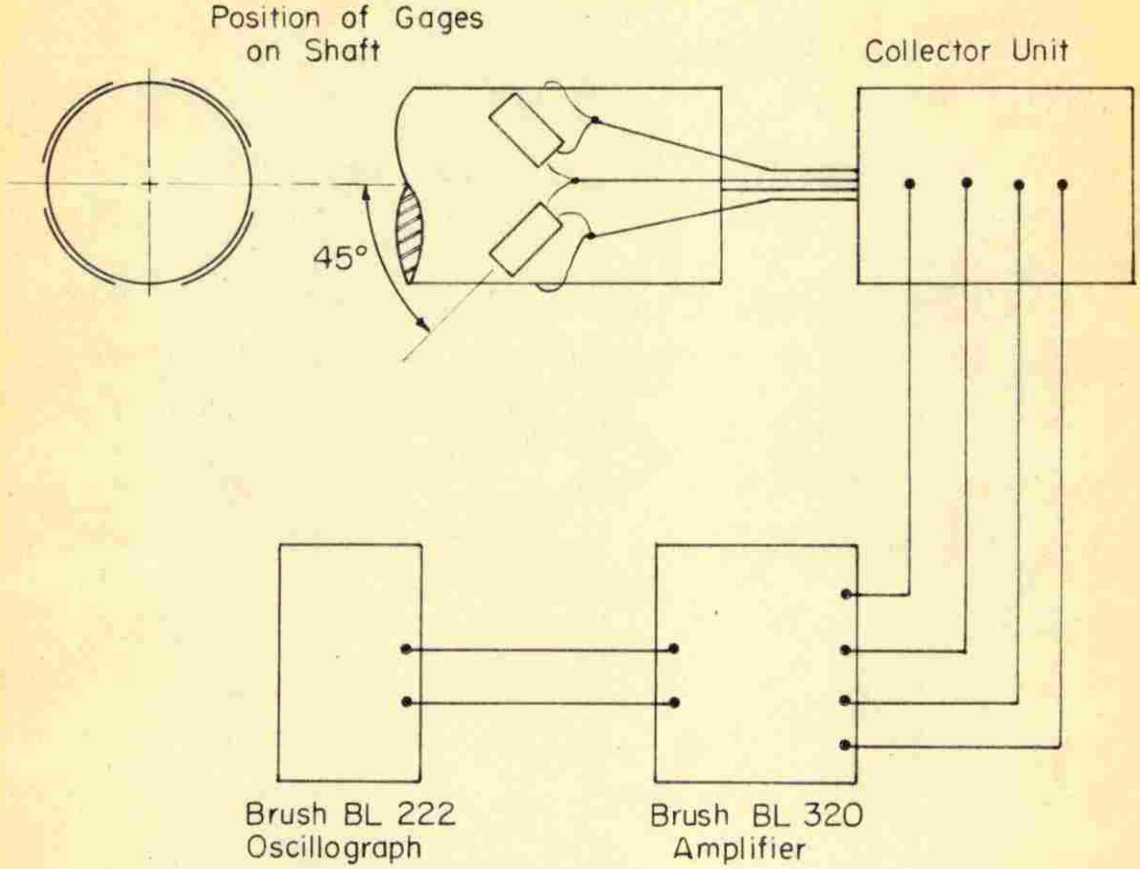


Figure 7. Arrangement of strain gages and instruments used for power measurement

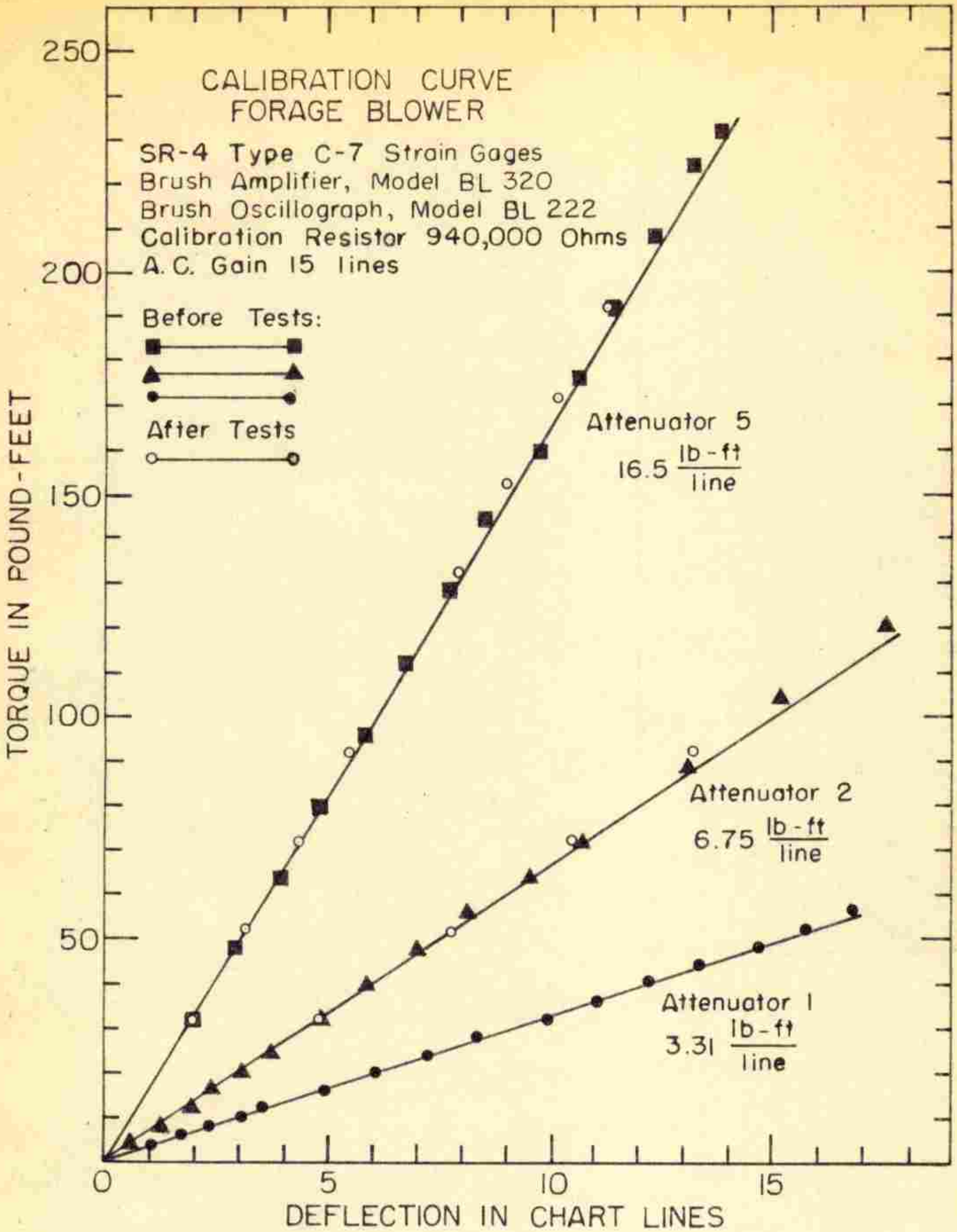


Figure 8. Calibration curves for determination of torque from oscillograph charts

means of throttling the air flow in order to vary the air delivery at constant blower speed had to be devised.

A thin-plate orifice flow-meter was made available for the investigation by the John Deere Ottumwa Works. The orifice diameter was 4.40 inches, with the pipe diameter being 8 inches. The meter was of the vena contracta type, with the upstream pressure tap one pipe diameter from the orifice and the downstream tap 0.61 diameters from the orifice. The orifice coefficient of discharge, calculated by the John Deere Ottumwa Works, was 0.64.

The orifice meter was placed in the blower pipe with the orifice plate located 9 feet-8 inches downstream from the pipe transition. This was 14-1/2 pipe diameters from any upstream change in cross section. The discharge end of the pipe was 13 feet-1 inch downstream from the orifice. A U-tube manometer was connected across the orifice and registered difference in static pressure in inches of water. From this differential pressure, the quantity of air per unit time in cfm (cubic feet per minute) was calculated by appropriate formulae. See Appendix D. Figure 9 shows the orifice meter in position in the blower pipe.

Static pressure in the pipe just above the pipe transition was also measured. A pressure tap was installed in the pivot section 3 inches above the transition upper flange. This tap was connected to one side of a U-tube manometer and registered static pressure in inches of water above atmospheric pressure.

Throttling of the air flow was accomplished by an adjustable sheet metal gate at the discharge end of the blower pipe. By using slots in the gate and by swinging the gate on one bolt, any desired degree of throttling could be obtained. The gate is shown in Figure 10.

Air-Flow Tests

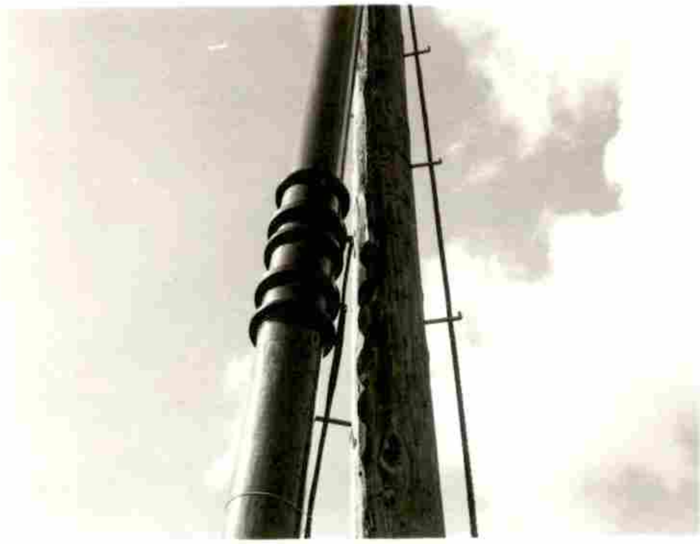
For the air-flow tests, the pipe height above the blower center was 27 feet-10 inches. The feed opening was 17-1/4 by 14 inches for all tests. The test set up is shown in Figures 11 and 12.

The first group of tests were run with free air delivery; that is, no throttling of the air flow. With the tip-clearance set, runs were made at nominal blower speeds of 500, 600, 700, 800, 900, and 960 rpm (revolutions per minute). At each speed, four torque records were made on the Brush recorder and four manometer readings were taken. Since 1000 rpm was considered top blower speed and 960 rpm could be reached at full throttle, it was not considered worthwhile to change to a smaller blower pulley in order to reach 1000 rpm.

The procedure described above was carried out for the following tip-clearances: 1/32, 1/16, 1/8, 1/4, 3/8, and 1/2 inch. The change in peripheral speed of the paddles over this range of tip-clearances was too small to take into account when adjusting blower speed.

Figure 9. Orifice meter in blower pipe

Figure 10. Air-flow throttling gate



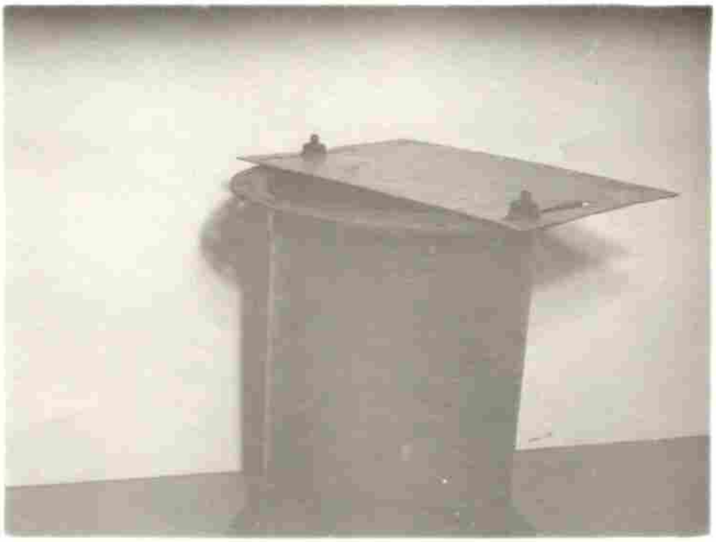
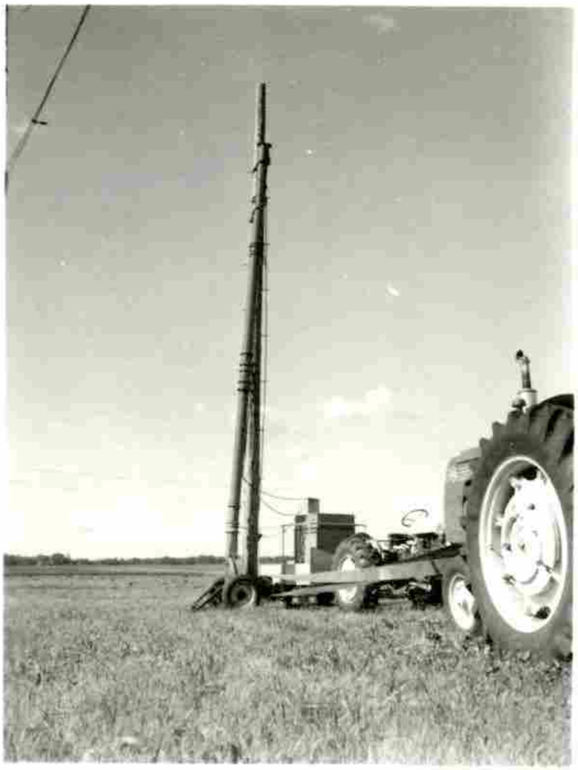
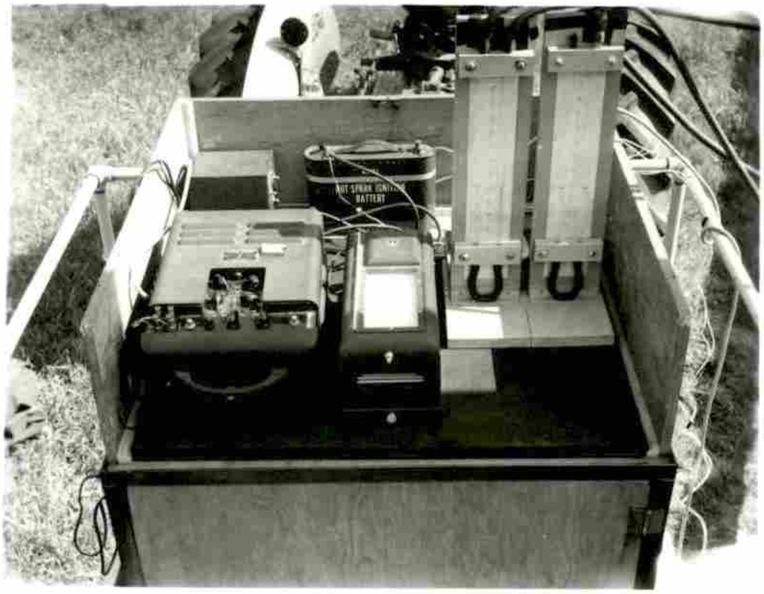


Figure 11. Air-flow test set-up

Figure 12. Instruments for power measurement and manometers for air-flow measurement





The second group of tests were run with throttled airflow. For each tip-clearance and speed, runs were made with the discharge end of the pipe 1/2 open, 1/4 open and closed. The same tip-clearances were used. Only three speeds - 600, 800, and 960 rpm - were used, as the first group of tests had definitely established a fixed relationship between speed of the blower and the quantity and static pressure of the air. The number of observations at each speed was also reduced to three, due to the small variation in reading the manometers.

The torque and speed used to calculate power for a given condition were the average values of torque and speed for the observations at that condition. Likewise, the quantity of air was calculated from the average differential pressure head observed and the static pressure is the average of the observations. Torque was obtained from the chart by planimetering the area under a length of the curve, then determining the average ordinate in millimeter lines. This average ordinate was converted to average torque in pound-feet by use of the calibration curve. Speed used in power calculations was determined from the revolution marks made on the chart as described in the section on instrumentation. Calculation of power and of quantity of air are respectively covered in Appendices C and D.

Forage Tests

Preliminary tests

A group of preliminary tests were conducted in order to determine the degree of effect, on power-requirement and capacity, of variations in tip-clearance and blower speed and to evaluate equipment and test procedures. Tip-clearances of 1/32, 1/16, 1/8, 1/4, 3/8 and 1/2 inch and speeds of 600, 750, 850, and 950 rpm were selected for the tests. Tests were run at all of the six tip-clearances for the two extreme speeds, but only at 1/32 and 1/2 inch for the two intermediate speeds. The pipe height above the blower center for the tests was 26 feet-6 inches.

Description of tests. Chopped alfalfa hay with a moisture content of 20 per cent and a theoretical length of cut of 3 inches was used for the tests. The desired feed rate was obtained by using the required combination of hay-load per unit length and linear speed on the conveyor. By distributing, for example, 100 pounds of hay uniformly over the 20-foot length of conveyor, a load of 5 pounds per foot was obtained. With a known load, a range of feed rates were available by adjusting the throttle on the tractor which was used to drive the conveyor.

With the tip-clearance set, the feed conveyor was placed in position and its speed regulated. Conveyor speed was measured by the use of a tachometer on the conveyor-chain drive shaft, with the relationship between linear speed and the shaft speed previously determined. When the speed was adjusted, the power-take-off was disengaged and the throttle left in position. Hay was placed in a canvas sheet and weighed on a dynamometer-type scale. The hay was then distributed as uniformly as possible on the conveyor. Blower speed was set by adjusting the tractor throttle and measuring blower shaft rpm with a tachometer. The Brush oscillograph and amplifier were prepared by balancing, zeroing, setting the attenuator and setting the chart speed. The test set-up is shown in Figures 13 through 16.

As the hay was fed into the blower, torque and speed records were made by the oscillograph. During the run, the conveyor speed was again checked with the tachometer. Feed rate was then adjusted and additional runs made, for each tip-clearance and speed, until the apparent capacity was reached. The so-called capacity was the maximum feed rate which in the author's opinion the blower could deliver without the pipe plugging or blockage occurring in the housing. Visual observation was made possible by the "Plexiglass" sides on the blower housing. Plugging of the pipe and subsequent wedging of material in the housing were frequent occurrences in the effort to judge capacity. Power calculation is covered in Appendix C.

Figure 13. Forage test set-up

Figure 14. Hay being weighed prior to placement on feed conveyor

Figure 15. Blower, feed conveyor, and conveyor drive

Figure 16. Loaded conveyor and blower

<u>Results of preliminary tests</u>. The test results, though quite useful in planning further tests, were not conclusive with regard to definite comparisons between tip-clearances. The results were not considered suitable for graphical presentation, but a summary is shown in Table 1. Complete data are shown in Appendix G. Only one relationship was definitely established from the test data; that was, that capacity increased with blower speed at all tip-clearances. Reasons for the lack of a basis for comparison in the data were:

1. there was a great deal of variation between the actual blower speed and the nominal blower speed in many cases;

2. there was no duplication of feed rates at different tip-clearances;

3. the determination of capacity was too arbitrary;

4. there was no replication of runs.

Any quantitative discussion of these results will be omitted, as results of later tests provide a much sounder basis for such discussion.

The preliminary tests did provide a general picture of the behavior of the blower with different tip-clearances. They indicated that the test facilities were adequate and that general test procedure was satisfactory. The results also pointed out the type of tests which were necessary to establish definitely the effect of tip-clearance on power-requirement and capacity.

Table 1. Summary of results of preliminary forage tests[a]

Tip-clearance (in.)	Nominal blower speed (rpm)	Observed speed (rpm)	Capacity (ton/hr)	Power (hp)	Hp-hr per ton
1/32	600	581	10.7	6.6	0.62
1/16		510	26.4	14.6	0.55
1/8		556	24.7	17.5	0.71
1/4		556	18.1	15.5	0.86
3/8		528	26.4	12.6	0.48
1/2		600	7.0	2.3	0.32
1/32	750	714	13.1	10.1	0.77
1/2		711	21.1	14.6	0.69
1/32	850	820	18.9	17.8	0.94
1/2		829	25.4	23.4	0.92
1/32	950	909	21.1	31.7	1.51
1/16		820	35.5	32.2	0.91
1/8		877	36.4	39.4	1.08
1/4		852	32.1	37.2	1.16
3/8		712	44.7	32.4	0.72
1/2		915	31.9	25.0	0.78

[a]Tests were on alfalfa hay with 20 per cent moisture content and 3-inch length of cut

Power requirement tests

The results of the preliminary tests indicated the need for holding feed rate and blower speed constant in order to determine the relation between tip-clearance and power-requirement. Three series of tests, involving two blower speeds and two feed rates, were run. At each tip-clearance setting, for each series, three runs were made in order to increase the reliability of the data. All tests were run with chopped alfalfa hay with a theoretical length of cut of 3 inches. The moisture content varied from 18 to 24 per cent.

For the first series of tests, a blower speed of 800 rpm and a feed rate of 20.6 tons per hour were selected. The hay moisture content was 18 per cent. Tests were run at tip-clearances of 1/32, 1/16, 1/8, 1/4, 3/8 and 1/2 inch. For each run the conveyor was loaded with 80 pounds of hay, giving a unit load of 4 pounds per foot. The conveyor drive-shaft speed was set at 156 rpm, resulting in a linear speed of 2.86 feet per second. The procedure for conduct of the test run was the same as was previously described for the preliminary tests. The height of the blower pipe above the center of the blower was 27 feet-6 inches for this and the two subsequent series of tests.

The second series of tests was run at the same blower speed, but at a feed rate of 14.5 tons per hour. The moisture content of the hay was 23 per cent. Tests were run at tip-clearances of 1/16, 1/4, 3/8 and 1/2 inch. The unit hay-load on the conveyor was again 4 pounds per foot. The linear speed of the conveyor was 2.01 feet per second.

For the third series of power-requirement tests, the blower speed was changed to 750 rpm, but the feed rate was maintained at 14.5 tons per hour. The moisture content of the hay was 24 per cent. Tip-clearances of 1/32, 1/16, 1/8, 1/4, 3/8 and 1/2 inch were used for the tests.

For each test, the Brush oscillograph chart furnished a record of torque and speed on the blower shaft. Power calculation is covered in Appendix C. Typical sections of oscillograph chart are shown in Figure 17.

Capacity tests

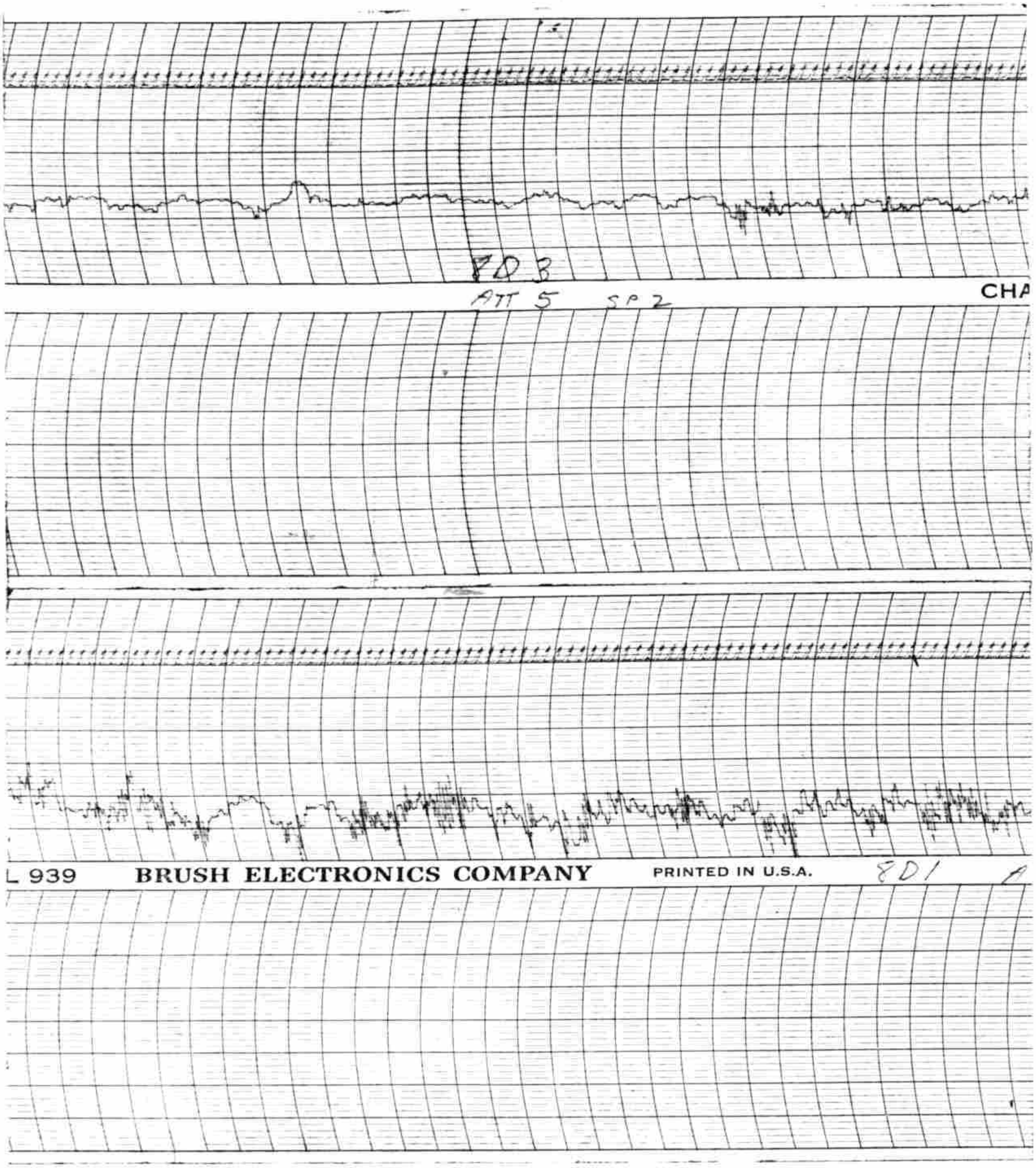
Based on observations and results of the preliminary tests it appeared that the logical way to determine capacity for a particular nominal speed was on the basis of an allowable speed-drop. This would not necessarily have been true if a power source had been available which would maintain a nearly constant speed with increased feed rate.

A nominal blower speed of 850 rpm was selected for running the capacity tests. A maximum allowable speed-drop of 10 per cent was decided upon. Capacity was defined as that feed rate which produced a drop in blower speed of between seven and ten per cent of the nominal speed.

Hay for the tests was chopped alfalfa with a 3-inch

Figure 17. Typical sections of oscillograph
chart from forage tests

(Base line is center of upper half of each chart)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length of cut and a moisture content of 24 per cent. The pipe height was the same as for the power-requirement tests. Tests were run at tip-clearances of 1/16, 1/8, 1/4, 5/16, 3/8 and 1/2 inch. Since the power-requirements had been well established in prior tests and to minimize the time required for the tests, no power measurements were taken during the capacity tests.

The blower speed was adjusted as measured by the tachometer to 850 plus 10, minus 0 rpm. As the hay was fed in, the speed was recorded by the event marker on the oscillograph chart. From the known chart speed, the blower speed during the run could be calculated. If the blower speed did not drop to between 765 and 790 rpm, the feed rate was adjusted accordingly. When the feed rate was found which produced the desired speed drop, a second run was made at that feed rate as a check. If the results were consistent, the feed rate was considered capacity for that tip-clearance.

RESULTS AND DISCUSSION

Air-Flow Tests

The measurement of quantity of air per unit time and of static pressure were considered satisfactory. A summary of results is shown in Tables 2 and 3. Complete data are shown in Appendix C. All values of static pressure, quantity of air and power are corrected to the nominal speed. The calculations are shown in Appendix D. It will be noted that tip-clearance in the range tested had a negligible effect on both static pressure and quantity of air.

Fan characteristic curves, showing the variation of power and static pressure with changes in quantity of air at constant speed, for different tip-clearances, were the desired results from the air-flow tests. However, the power-requirement data were unsatisfactory due to experimental error in the measurement of torque. The resulting characteristic curves were meaningless. The error in torque measurement was mainly due to the limitations of the strain gage instrumentation as applied to this particular installation.

When operating as a fan, a very low torque, in the neighborhood of 5 to 10 pound-feet, was developed on the blower shaft. As the blower shaft was 1-1/2 inches in diameter, the torsional strain was of a small magnitude, resulting in a correspondingly small deflection of the oscillograph pen. An

Table 2. Summary of results of air-flow tests (volume in cubic feet per minute)

Blower speed (rpm)	Degree of throttling	Tip-clearance (in.)					
		1/32	1/16	1/8	1/4	3/8	1/2
600	0	412	404	395	386	398	393
	1/2	326	323	335	333	321	317
	3/4	208	200	206	200	202	193
800	0	541	542	538	520	533	527
	1/2	447	446	444	446	439	438
	3/4	275	273	279	275	276	263
960	0	644	650	645	634	639	635
	1/2	538	537	542	544	534	522
	3/4	332	332	330	326	333	325

Table 3. Summary of results of air-flow tests (static pressure in inches of water, gage)

Blower speed (rpm)	Degree of throttling	Tip-clearance (in.)					
		1/32	1/16	1/8	1/4	3/8	1/2
600	0	1.66	1.50	1.49	1.49	1.52	1.48
	1/2	1.52	1.54	1.51	1.51	1.55	1.49
	3/4	1.61	1.59	1.56	1.51	1.52	1.57
	Full	1.58	1.57	1.62	1.59	1.62	1.52
800	0	2.95	2.72	2.69	2.71	2.73	2.71
	1/2	2.82	2.83	2.76	2.72	2.80	2.69
	3/4	2.81	2.84	2.87	2.81	2.79	2.74
	Full	2.92	2.90	2.89	2.85	2.92	2.75
960	0	4.02	3.97	3.87	3.88	3.93	3.88
	1/2	4.04	4.03	4.00	4.03	3.98	3.84
	3/4	4.06	4.09	4.03	3.98	4.08	3.96
	Full	4.20	4.15	4.12	4.04	4.14	4.01

attenuator setting of 1, the most sensitive on the Brush amplifier, was unstable, apparently due to some outside electrical interference. On an attenuator setting of 2, the oscillograph pen deflection was less than two lines.

An inherent characteristic of the amplifier electrical circuits involved in zeroing the oscillograph pen is a drift with time. There was a necessary time lag between zeroing the instrument and running the test. During this and the ensuing time of the test, the zero might drift as much as one line. The drift was not regular, and it was impossible to know when it started or what its magnitude was at various points on the chart. This drift was such a large part of the total deflection, that any attempt to correct for it resulted in badly inconsistent torque values.

Since the power-requirement when operating in the normal speed range of the blower was only in the neighborhood of 1 or 2 horsepower and since the blower was not primarily a fan, it was felt that the importance of the fan characteristics was not such as to justify the trial of other methods of power determination.

Forage Tests

Power requirement

Experimental. Results of the three series of power-requirement tests are shown in Figure 18. The curves represent

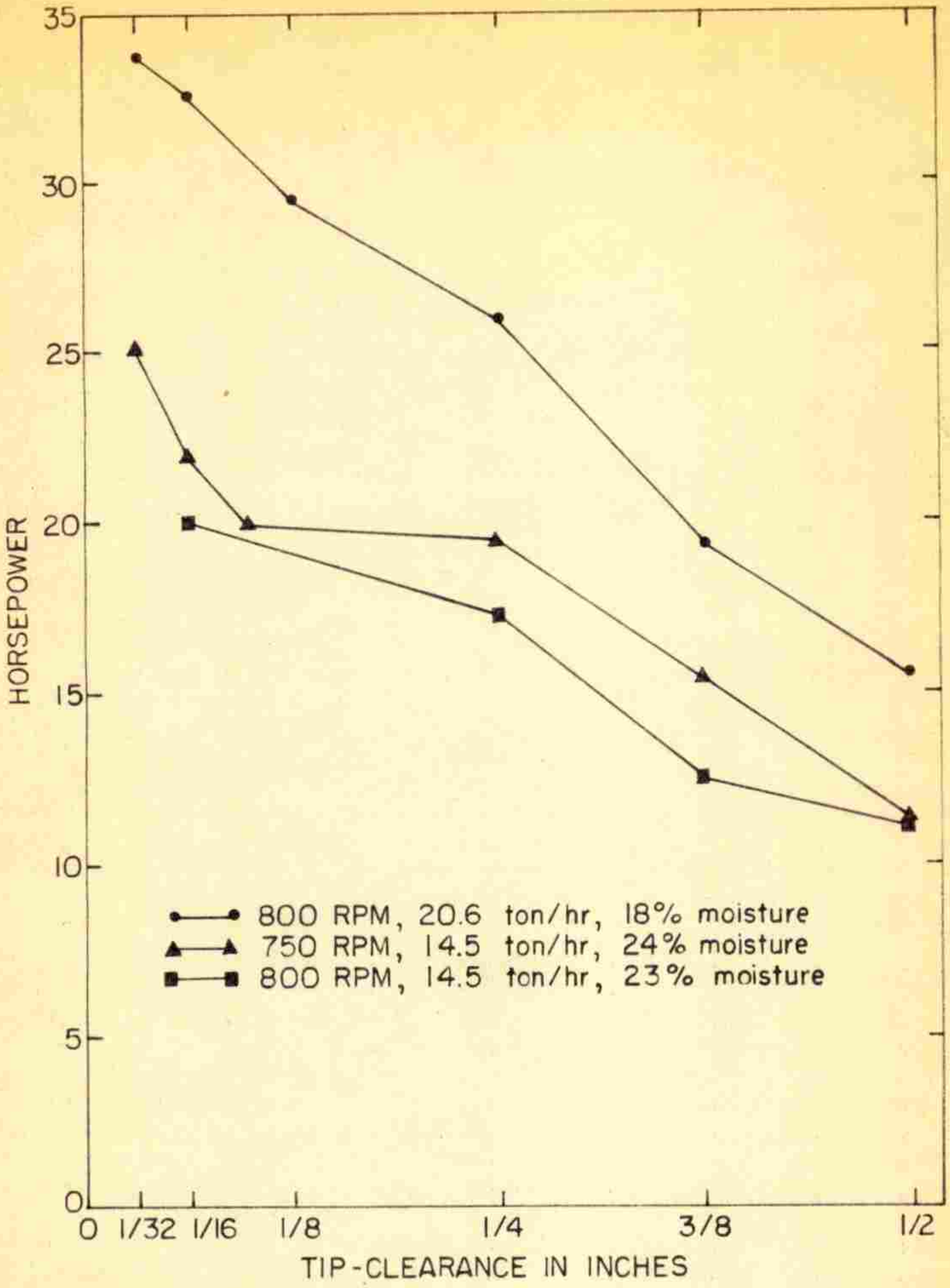


Figure 18. Power-requirement as a function of paddle tip-clearance for the forage blower delivering alfalfa hay with 3-inch length of cut

the effect of paddle tip-clearance on the power-requirement at two constant speeds and two constant feed rates. Each point is the average power for two or, in most cases, three runs. Complete data are presented in Appendix C.

Variation in the quantity of power determined for the individual runs at a given condition was due to several factors. Speed variation, though as much as 4 or 5 per cent below the nominal speed, was quite small between runs. Lack of homogeneity of the hay probably contributed a small amount to the differences. A major factor appeared to be the failure of the mass of hay to break apart in a uniform manner as it moved from the conveyor into the blower feed opening. This could cause a greater amount of carryover of hay on some runs than on others. There was some error in the recording of torque due to instrument limitations and, no doubt, some in chart analysis. The instrument limitations were previously discussed in the results of the air-flow tests.

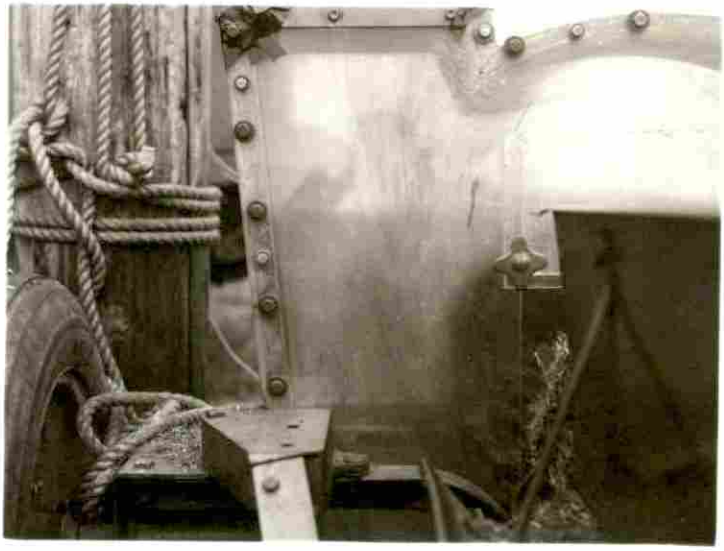
The results show clearly the reduction in power-requirement as the tip-clearance is increased from 1/32 to 1/2 inch. The changes in feed rate and in blower speed did not essentially alter the effect of tip-clearance on the power-requirement. The rate of change of power with tip-clearance was slightly higher at the higher feed rate.

The greatest single contribution to the power-requirement of this blower appears to be that due to interference of hay at the cutoff point and subsequent carryover involving considerable friction loss. The transparent sides on the blower housing permitted visual observation of the relatively large quantity of hay which was not released into the discharge pipe, but caught between the paddles and the cutoff point. See Figure 19. The sound of this violent action was also quite distinguishable. The energy absorption of this "hairpinning" action was no doubt significant. The hay, after passing the cutoff point, was partly wedged between the paddle tip and the housing, resulting in a frictional force which in the extreme case was sufficient to stall the blower. The importance of the relative magnitude of the power consumed in this manner will be supported by a theoretical analysis of power-requirement which is made below.

Carryover of material, as described above, was not restricted to the smaller tip-clearances, but appeared to be approximately the same for all. However, the larger tip-clearances allowed more space for this hay to pass the cutoff point and to be contained in as it was carried around the housing. This resulted in less energy absorption at the cutoff point and a smaller frictional force opposing the motion of the paddle. As the tip-clearance was increased, the sound of the cutoff point interference decreased and at 1/2 inch was barely audible, even though the carryover was clearly visible. The torque curve on the Brush recorder was considerably smoother at the larger tip-clearances.

It will be noted that the curve for a blower speed of 750

Figure 19. Hay-movement through blower-housing



rpm lies above the curve for 800 rpm, both at a feed rate of 14.5 tons per hour. It is believed that this is due to increased carryover of hay as a result of the decrease in the energy imparted to the hay. For this speed reduction of 50 rpm, a particle leaving a paddle tip would have 12.5 per cent less kinetic energy than at 800 rpm. It seems reasonable that, with hay coming into the blower at the same rate, there would be more interference at the cutoff point on the lower speed. If this were the case, the increase in power-consumption at the cutoff and by friction could account for the higher power-requirement at 750 rpm. One minor contributing factor could have been the 1 per cent higher moisture content of the hay blown at 750 rpm, which would be expected to increase the friction loss.

Theoretical analysis. The total power required by the blower may be considered as made up of three components: power for change in the kinetic energy of the material; power for overcoming the friction of the material against the housing; and, power for pumping air. That is,

Total power, $P = P_{KE} + P_F + P_A$.

Theoretical equations for P_{KE} and P_F in horsepower are derived in Appendix E. The equations are

$$P_{KE} = 1.29(10)^{-9}\, F\, D^2\, N^2 \quad \text{and}$$

$$P_F = 1.07(10)^{-8} \mu \alpha\, F\, R^2\, N^2$$

where F = feed rate in pounds per minute

D = diameter of the impeller in feet

N = blower speed in revolutions per minute

μ = kinetic coefficient of friction

α = angle of contact of material and housing in radians

R = radius of blower housing in feet

For P_A, the usual procedure in a theoretical analysis is to use the fan characteristic curve and assume that the power for pumping air remains constant when blowing material. As previously explained, the no-load data for this experiment were not suitable for plotting a fan characteristic curve. However, based on the large number of power calculations made, it is felt that a reasonable estimate of this power requirement can be made for purposes of this analysis.

Blevins and Hansen (1) determined kinetic coefficients of friction between steel and alfalfa at various moisture contents. The surface speed used in the determinations is not known, but a coefficient of approximately 0.25 was obtained for a moisture content in the neighborhood of 20 per cent. Schinke (16) determined a sliding coefficient of 0.5 for green, chopped alfalfa on steel at speeds of from 1000 to 6000 feet per minute. Richter (15) ran tests at speeds below 640 feet per minute for chopped straw. He recommends a coefficient of 0.30 for the chopped straw. The alfalfa hay used in this experiment would probably have a slightly higher coefficient than the chopped straw at the same surface speed. The surface speeds in this experiment were generally above 6000 feet per minute. A coefficient of 0.30 is probably a reasonable estimate and will be used for this analysis.

The angle of contact is generally assumed to be 90 degrees. This assumption considers that the material falls to the bottom of the housing and exerts a normal force on the housing band around to the tangent point. Actually, some of the hay is caught in an air-stream and never acts in the assumed manner. The assumption also does not account for any housing contact as a result of carryover.

Calculations will be made for a tip-clearance of 1/2-inch. The following values will be substituted:

F = 483 pounds per minute (14.5 tons per hour)

D = 2.75 feet (33 inches)

R = 1.42 feet (17 inches)

N = 800 revolutions per minute

μ = 0.30

$\alpha = \pi/2$ radians

$$P_{KE} = 1.29(10)^{-9}(483)\ (2.75)^2\ (800)^2$$

$$= 3.02 \text{ horsepower}$$

$$P_F = 1.07(10)^{-8}(0.30)\ (\pi/2)\ (483)\ (1.42)^2\ (800)^2$$

$$= 3.16 \text{ horsepower}$$

P_A (estimated) = 2.00 horsepower

$$P = P_{KE} + P_F + P_A = 3.02 + 3.16 + 2.00$$

$$P = 8.18 \text{ horsepower}$$

For the above conditions, the experimental power-requirement was 11.1 horsepower. This is the test condition at which the power due to cutoff-point interference and carryover is a minimum. Recognizing the limiting assumptions in the theoretical analysis (see Appendix E) the difference of 2.9 horsepower should give some indication of the magnitude of the power consumed by the interference at the cutoff point and the carryover of material. The theoretical power-requirement for the other tip-clearances would change very little due to change in radius of the impeller.

Blower capacity

Results of the capacity tests are shown in Figure 20. It is important to realize that the plotted values do not represent the absolute capacity of the blower at that tip-clearance and speed. The values are rather feed rates which meet a specific definition of capacity. That is, capacity was defined as that feed rate which produced a drop in blower speed of between seven and ten per cent of the nominal speed.

The speed-drops for the four larger tip-clearances were, by coincidence, all at the lower limit of seven per cent. The drop for 1/8-inch was just below eight per cent, while the drop for 1/16-inch was the full ten per cent. Thus the difference between capacity at 1/16 and at 1/8-inch would have been somewhat less had finer limits been placed on the speed-drop. However, the feed rate which is shown as capacity for 1/16-inch actually plugged the blower on one run at 1/8-inch.

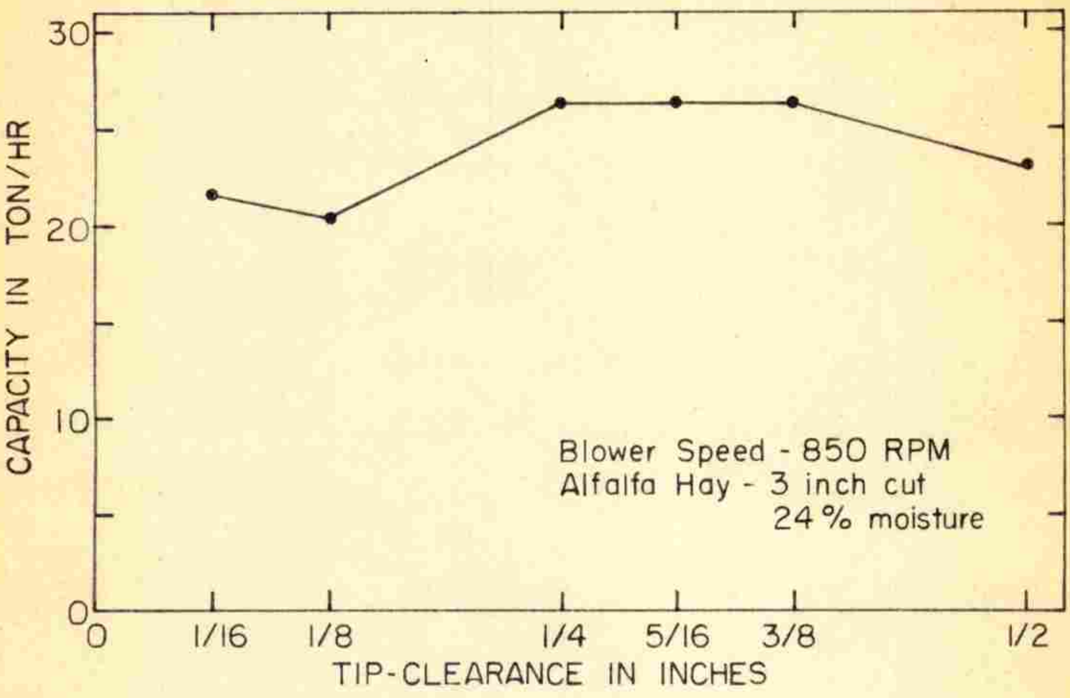


Figure 20. Forage blower capacity as a function of paddle tip-clearance

It will be noted that the highest capacity was obtained for tip-clearances of 1/4, 5/16, and 3/8 inch. In the preliminary tests, where speed drop was not considered and where capacity was judged by observation, a 3/8-inch setting delivered at the highest rate of 44.7 tons per hour while the speed dropped from 950 to 712 rpm.

Unloading of the blower paddles

Examination of Figure 19 sheds interesting light on a commonly used theoretical analysis of paddle unloading originated by Raney (14). A similar analysis is made in Appendix F. The assumptions usually made are that:

1. at the outlet tangent point, approximately on the horizontal center line, each particle of material is at rest with respect to the paddle;

2. unloading will begin as the paddle passes the outlet tangent point, due to centrifugal force; and

3. frictional forces may be neglected.

From the solution of the equation of motion of a particle, one of the resulting relationships gives the minimum radius of the position of a particle at rest on the paddle which will not result in interference at the cutoff point of a particular size blower. If all the assumptions were valid and if all of the material were fed onto a fixed portion of the paddle, it would be a simple matter to design for no interference. However, it is evident from the flow paths visible in Figure 19 that such is not the case in a blower with a commonly positioned feed opening. Much of the material never moves to the outer portion of the paddles. Material of higher moisture content would probably be better in this respect due to a higher specific weight.

Limitations of the Study and Suggestions for Future Research

This study was conducted with only long-chopped alfalfa hay with low moisture-content. The effects of paddle tip-clearance in handling green materials and other lengths of cut might conceivably be quite different. Further tests of different crops and crop conditions are suggested.

The pipe height was constant for these forage tests and was not representative of the maximum elevations encountered in practical situations. There is no definite indication of the limitation imposed by tip-clearance on a particular delivery elevation. There was no difficulty in delivering approximately thirty feet at any tip-clearance for blower speeds above 750 rpm. Tests are needed to establish the relationship of tip-clearance to elevating ability.

Conditions did not allow tests to be run in which the tip-clearance at the cutoff point was varied independently of the clearance around the housing. Since such a large part of the blower power-requirement appears to be connected with interference at the cutoff point and since tip-clearance does have considerable effect on the power-requirement, it definitely seems desirable to consider the cutoff tip-clearance independently.

One of the difficulties encountered in running tests which were suitable for comparisons, was that of maintaining a constant blower speed. A belt drive is definitely undesirable from this standpoint. However, the limitations of the centrifugal-type governor on the tractor engine was the chief source of speed variation. It is suggested that for future studies of this nature, consideration be given to a power source and drive system which will maintain close speed control over a wide torque range.

This study has brought into focus the large amount of power lost due to poor paddle unloading characteristics in a blower of this design. Though the blower used was slightly modified, it was fairly representative of many production blowers. There appears to be a need for some fundamental studies of housing and paddle configuration with relation to discharge characteristics. Closely related to this would be work on improved placement of incoming material.

SUMMARY AND CONCLUSIONS

The forage blower is the most common device for elevating chopped forages into storage structures. It is a simply constructed, relatively low-cost machine with a rather high capacity. The general operating performance of the blower, however, is not satisfactory for some crop conditions and its efficiency is very low. Most present-day blowers require at least a two-plow tractor, or its equivalent, for a power source.

Of the large number of variables which affect to some degree the performance of a forage blower, paddle tip-clearance is known to be a very important one. There is little evidence to justify presently used tip-clearance settings. No data are available on the effect of varying the paddle tip-clearance over a reasonable range.

This study seeks to determine the effect of paddle tip-clearance on the power-requirement, capacity, and over-all operation of the forage blower for elevating chopped hay. A John Deere Model 50 blower was modified to form a circular housing with transparent plastic sides. The tip-clearance of the four straight paddles was made adjustable. SR-4 electrical resistance strain gages, bonded to the blower shaft, were used in conjunction with a Brush recording oscillograph for measurement of torque.

Air-flow tests were conducted in an effort to determine fan characteristic curves for various tip-clearances and speeds. Quantity of air per unit time was measured by means of a thin-plate orifice meter. Static pressure just above the pipe transition was determined. Power measurements for the air-flow tests were unsatisfactory due to the small magnitude of the measurement and a large experimental error in instrumentation. The series of tests covered six speeds from 600 to 960 revolutions per minute and six tip-clearances from 1/32 to 1/2 inch. In these ranges, paddle tip-clearance had a negligible effect on both static pressure and quantity of air.

All forage tests were run with chopped alfalfa hay with a 3-inch length of cut and a moisture-content range of 18 to 24 per cent. After a group of preliminary tests to determine the degree of effect of variations in paddle tip-clearance and blower speed and to evaluate equipment and test procedures, power-requirement tests were run at different tip-clearances with a constant blower speed and feed rate. Three such series were run, involving two blower speeds and two feed rates. The results of each series of tests showed a significant decrease in power-requirement with increase in tip-clearance.

A final group of tests were conducted to determine the variation of capacity with tip-clearance. The tests were run at a blower speed of 850 revolutions per minute. Capacity was defined as that feed rate which caused a drop in speed of from seven to ten per cent. The maximum capacity was obtained for tip-clearances of 1/4, 5/16, and 3/8 inch.

It was observed that the greatest single contribution to the power-requirement of the blower appeared to be that due to interference of hay at the cutoff point and subsequent carryover. This observation was supported by a theoretical analysis of power-requirement, which was substantially lower than the experimental results. The effect of the carryover of material was much greater for the smaller tip-clearances.

A theoretical analysis of paddle unloading was presented. The theoretical analysis was compared with photographic evidence of the actual material flow paths present in the test blower.

From this study of a forage blower with a circular housing and straight paddles, handling low-moisture, long-chopped alfalfa, the following conclusions were drawn:

1. Paddle tip-clearance is definitely a critical factor in the performance of the blower.

2. Tip-clearance had a negligible effect on quantity of air and on static pressure in the range of clearances and speeds investigated.

3. In the normal operating speed range, power-requirement for a given feed rate decreased with increased tip-clearance in the range from 1/32 to 1/2 inch.

4. The largest single contribution to the power-requirement of the blower was that due to cutoff-point interference and subsequent carryover of hay. This source of power consumption was much greater at the smaller tip-clearances,

accounting for as much as a 100 per cent increase in power-requirement for 1/32-inch as compared to 1/2-inch tip-clearance.

5. Maximum capacity for the blower tested was at a tip-clearance in the range of 1/4 to 3/8 inch.

6. Based on visual observation, power requirement, and capacity, a tip-clearance of 3/8 inch appears most desirable.

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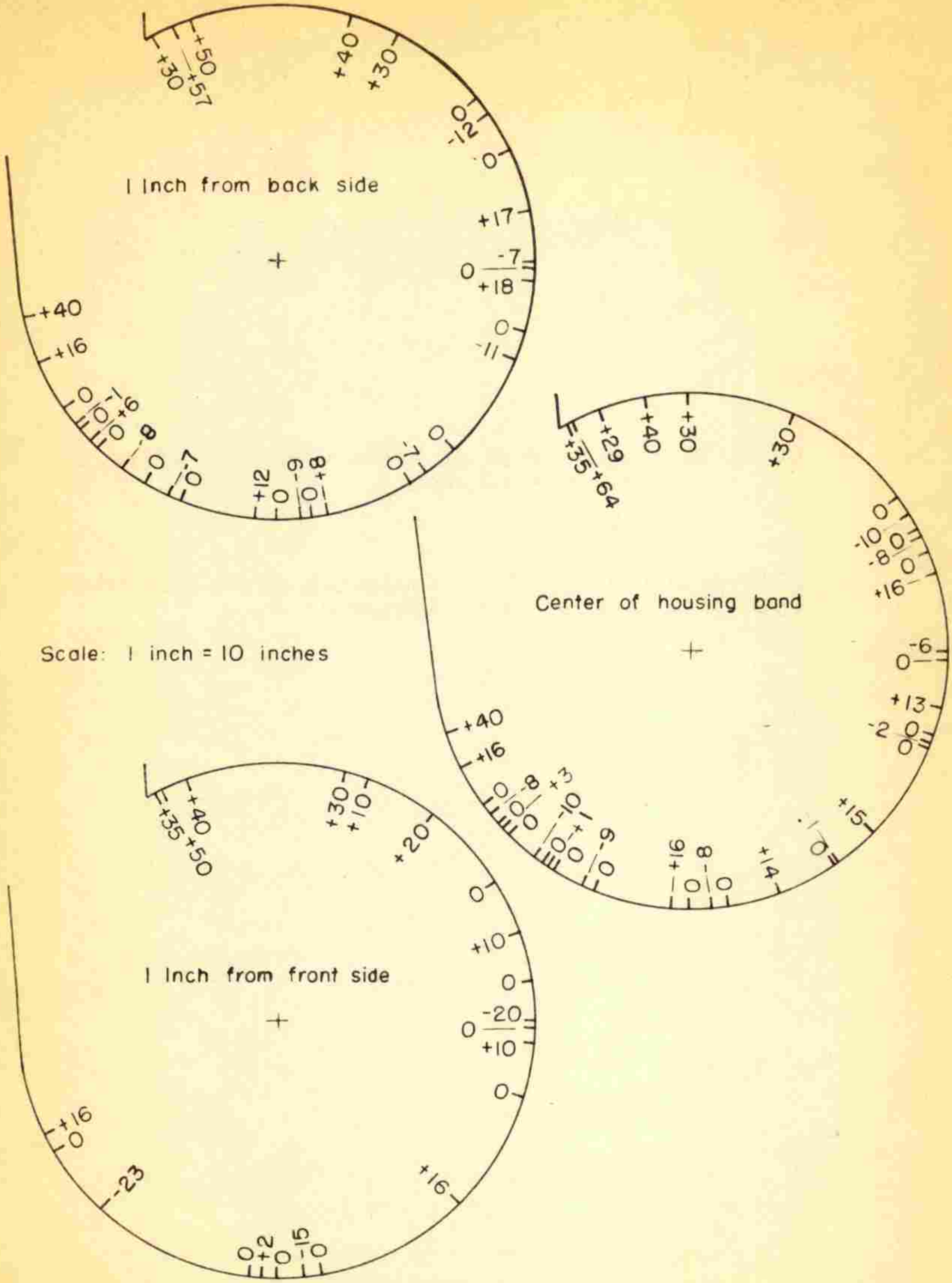
APPENDICES

Appendix A:

Blower-Housing Tolerances

Figure 21. Tolerances of blower-housing radius at three sections

(Dial gage readings in thousandths of an inch with bottom of housing as a reference)


1 Inch from back side
Center of housing band
Scale: 1 inch = 10 inches
1 Inch from front side

Appendix B:

Calibration Data

Table 4. SR-4 strain gage calibration data[a]

Weight (lb)	Torque (lb-ft)	Lines deflection Attenuator 1	Attenuator 2	Attenuator 5
	Before tests - Torque arm = 2 feet			
1.9	3.8	1.0	0.6	
2.9	5.8	1.7		
3.9	7.8	2.3	1.2	
4.9	9.8	3.0		
5.9	11.8	3.5	1.9	
7.9	15.8	4.9	2.4	
9.9	19.8	6.0	3.1	
11.9	23.8	7.2	3.7	
13.9	27.8	8.3		
15.9	31.8	9.9	4.9	
17.9	35.8	11.0		
19.9	39.8	12.2		
21.9	43.8	13.3		
23.9	47.8	14.6		
25.9	51.8	15.7		
27.9	55.8	16.7		
	Before tests - Torque arm = 4 feet			
7.9	31.6		4.9	1.9
9.9	39.6		5.9	
11.9	47.6		7.0	2.9
13.9	55.6		8.1	
15.9	63.6		9.5	3.9
17.9	71.6		10.7	
19.9	79.6			4.8
21.9	87.6		13.0	
23.9	95.6			5.8
25.9	103.6		15.1	
27.9	111.6			6.7
29.9	119.6		17.4	

[a]Calibration was on a Brush BL 320 amplifier and BL 222 oscillograph using a 940,000 ohm calibration resistor and AC gain of 15 lines.

Table 4. (Continued)

Weight (lb)	Torque (lb-ft)	Lines deflection Attenuator 1	Attenuator 2	Attenuator 5
31.9	127.6			7.7
35.9	143.6			8.5
39.9	159.6			9.7
43.9	175.6			10.6
47.9	191.6			11.3
51.9	207.6			12.3
55.9	223.6			13.2
57.9	231.6			13.8
	After tests - Torque arm = 4 feet			
7.9	31.6		4.8	1.9
12.9	51.6		7.7	3.1
17.9	71.6		10.4	4.3
22.9	91.6		13.1	5.4
32.9	131.6			7.9
37.9	151.6			9.0
42.9	171.6			10.1
47.9	191.6			11.2

Appendix C:

Calculation of Power

Let A = area under a length, L, of the deflection vs. time curve on the oscillograph chart, in square centimeters

L = length of oscillograph chart in centimeters

D = average ordinate of deflection-time curve in millimeter-lines

C = calibration constant in pound-feet of torque per line deflection

T = average torque in pound-feet

N = blower speed in revolutions per minute

S = chart speed in millimeters per minute

d = chart distance in millimeters

n = number of revolutions recorded in distance, d

P = power in horsepower

$$D = 10 \frac{A}{L}$$

$$T = CD$$

$$N = \frac{Sn}{d}$$

$$P = \frac{2 \pi T N}{33{,}000} = \frac{T N}{5250}$$

Appendix D:

Calculations for Air-Flow Data

Derivation of the formula for quantity of air per unit time

Air at 70 degrees, F. and a barometric pressure of 29.92 inches of mercury will be considered standard air. All observations will be reduced to standard air for basis of comparison.

Let V = velocity of air in the orifice meter in feet per minute

V_s = velocity of standard air in feet per minute

Q = quantity of air flow at standard conditions in cubic feet per minute

C = orifice coefficient of discharge (0.64)

g = acceleration of gravity in feet per second per second

H = differential static pressure across the orifice in feet of air

p = differential static pressure across the orifice in inches of water

d_w = specific weight of water at 70 degrees, F. in pounds per cubic foot

d_a = specific weight of ambient air in pounds per cubic foot

d_s = specific weight of standard air in pounds per cubic foot

A = area of the orifice in square feet

D = orifice diameter in inches (4.40)

The velocity of air through the orifice is given by

$$V = 60C\sqrt{2\ g\ H}$$

But $$H = \frac{p}{12}\ \frac{d_w}{d_a}$$

Therefore $$V = 60C\sqrt{\frac{g\ p}{6}\ \frac{d_w}{d_a}}$$

$$= 88.9\sqrt{p\ \frac{d_w}{d_a}}$$

According to <u>Fan Engineering</u> (11, p.99), for a given pressure, the velocities of air at two different specific weights vary inversely as the square root of the specific weights. That is,

$$\frac{V_s}{V} = \sqrt{\frac{d_a}{d_s}}$$

Then $$V_s = 88.9\sqrt{p\ \frac{d_w}{d_a}\ \frac{d_a}{d_s}} = 88.9\sqrt{p\ \frac{d_w}{d_s}}$$

$$d_w = 62.30$$

$$d_s = 0.07495$$

Therefore $$V_s = 2566\sqrt{p}$$

By definition, $$Q = AV_s = \frac{\pi\ D^2}{(4)\ (144)}\ 2566\sqrt{p}$$

$$Q = 271\sqrt{p}$$

Corrections of air flow data to nominal speed

In calculations to follow, subscript o indicates the observed value of a quantity and subscript c indicates the corrected value. According to Fan Engineering (11, p.261), for purposes of speed correction, volume varies directly as the speed, static pressure as the square of speed, and power as the cube of speed. Therefore,

$$\frac{Q_c}{Q_o} = \frac{N_n}{N_o} \qquad \text{where } N_n \text{ is the nominal speed,}$$

$$\frac{p_c}{p_o} = \left(\frac{N_n}{N_o}\right)^2 \qquad \text{and}$$

$$\frac{P_c}{P_o} = \left(\frac{N_n}{N_o}\right)^3$$

Appendix E:

Derivation of Theoretical Equations for Friction and Kinetic Energy Power-Requirements

<u>Derivation of equation for friction horsepower</u>

In the following analysis the assumption is made that frictional forces are acting only at the blower housing periphery. The symbols used are:

P_F = power to overcome friction, in horsepower

f_n = normal force on housing, in pounds

μ = sliding coefficient of friction

R = radius of blower housing, in feet

ω = angular velocity, in radians per minute

N = blower speed, in revolutions per minute

W = weight of material, in pounds per revolution

M = mass of material, in slugs per revolution

g = acceleration due to gravity, in feet per second per second

F = feed rate, in pounds per minute

α = angle of contact, in radians

$$P_F = \frac{\mu f_n\left(\frac{\alpha}{2\pi}\right)(2\pi R)\,N}{33{,}000} = \frac{\mu f_n R N \alpha}{33{,}000} \qquad (1)$$

By definition, $f_n = M R\omega^2 = \frac{W}{g} R\,(2\pi N)^2$

$$W = \frac{F}{N}$$

$$f_n = \frac{4\pi^2 F R N^2}{gN} = \frac{4\pi^2 F R N}{(32.2)(3600)}$$

$$= 3.52(10)^{-4} F R N \qquad (2)$$

Substituting (2) in (1),

$$P_F = \frac{3.52\ (10)^{-4}\ \mu\ \alpha\ F\ R^2\ N^2}{33{,}000}$$

$$= 1.07\ (10)^{-8}\ \mu\ \alpha\ F\ R^2\ N^2$$

Derivation of equation for kinetic energy horsepower

The assumptions are made that:

1. the kinetic energy of the material entering the blower is zero;
2. The final velocity of the material is equal to the peripheral velocity of the blower paddles.

The symbols used are:

P_{KE} = power to change kinetic energy, in horsepower

ΔKE = change in kinetic energy, in foot-pounds

D = diameter of the impeller, in feet

N = blower speed, in revolutions per minute

W = weight of material, in pounds per revolution

M = mass of material, in slugs per revolution

F = feed rate, in pounds per minute

V = peripheral velocity, in feet per minute

t = time, in minutes

g = acceleration due to gravity, in feet per second per second

$$\Delta KE = \frac{1}{2} M V^2 - 0 = \frac{W V^2}{2g}$$

By definition, power $= \frac{\Delta KE}{t}$

then, $P_{KE} = \frac{W V^2}{2 g t(33{,}000)}$

but, $\frac{W}{t} = F$ and $V^2 = (\pi D N)^2$

So, $P_{KE} = \frac{F(\pi D N)^2}{(2)(32.2)(3600)(33{,}000)}$

$$= 1.29 (10)^{-9} F D^2 N^2$$

Appendix F:

Theoretical Analysis of Paddle Unloading

The assumptions are:

1. at the outlet tangent point, each particle on the paddle will be at rest with respect to the paddle;
2. unloading will begin as the paddle passes the outlet tangent point;
3. frictional forces will be negligible between the paddle and the material.

Reference to Figure 22 will serve to define and locate the quantities used in the analysis.

We consider the forces acting on a particle moving radially on the paddles at some radius r between r_o and R. Summing forces in the radial direction,

ΣF_r = ma where m is mass and a is acceleration

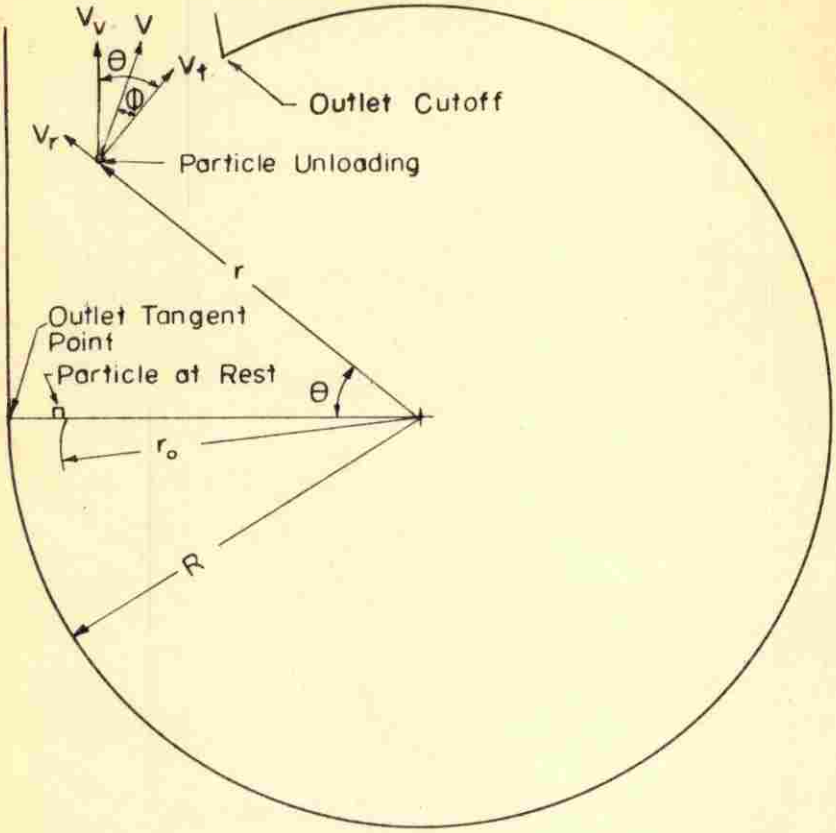
$mr\omega^2 = m \frac{d^2r}{dt^2}$ where t is time

$$\frac{d^2r}{dt^2} - r\omega^2 = 0 \qquad (1)$$

The general solution of equation (1) is

$$r = c_1 e^{\omega t} = c_2 e^{-\omega t} \qquad (2)$$

The initial conditions may be used to evaluate the constants, c_1 and c_2 in equation (2).



V_r = Radial Velocity
V_t = Tangential Velocity

V = Resultant Velocity
V_v = Vertical Component Velocity

Figure 22. Theoretical analysis of blower-paddle unloading

(a) when $t = 0$, $r = r_o$ and

(a') $r_o = c_1 + c_2$

(b) when $t = 0$, $\frac{dr}{dt} = 0$

$$\frac{dr}{dt} = c_1\omega e^{\omega t} - c_2\omega e^{-\omega t} = 0$$

$$\omega(c_1 - c_2) = 0$$

since $\omega \neq 0$,

(b') $$c_1 - c_2 = 0$$

thus $c_1 = c_2 = \frac{r_o}{2}$ and equation (2) becomes

$$r = \frac{r_o}{2}\,(e^{\omega t} + e^{-\omega t}) = r_o\,\left(\frac{e^{\omega t} + e^{-\omega t}}{2}\right) \tag{3}$$

By definition, $\frac{e^{\omega t} + e^{-\omega t}}{2} = \cosh \omega t$

and $\omega t = \theta$

Therefore, $r = r_o \cosh \theta$ (4)

Equation (4) gives the position of the particle after it has moved through an angle, θ.

The magnitude and direction of the velocity will now be determined. The angle between the resultant velocity and the tangential component of velocity is ϕ.

Then $\tan\phi = \frac{V_r}{V_t}$

$$V_r = \frac{dr}{dt} = \frac{dr}{d\theta}\,\frac{d\theta}{dt} = \omega\,\frac{dr}{d\theta}$$

Since $r = r_o \cosh\theta$,

$$\frac{dr}{d\theta} = r_o \sinh\theta \quad \text{and}$$

$$V_r = r_o\omega \sinh\theta \tag{5}$$

$$V_t = r\omega = r_o\,\omega \cosh\theta \tag{6}$$

$$\frac{Vr}{Vt} = \frac{\sinh\theta}{\cosh\theta} \quad \text{or}$$

$$\tan\phi = \frac{\sinh\theta}{\cosh\theta} \tag{7}$$

Equation (7) gives the relation of the angle which the resultant velocity makes with the tangential component, to the angle through which the particle has moved.

The vertical component of velocity,

$$V_v = V \cos(\theta - \phi)$$

$$V = \frac{V_t}{\cos\phi} = V_t \sec\phi \quad \text{and}$$

$$V_v = V_t \sec\phi \cos(\theta - \phi)$$

$$\sec\phi = \sqrt{1 + \tan^2\phi} = \sqrt{1 + \left(\frac{\sinh\theta}{\cosh\theta}\right)^2}$$

$$V_v = V_t \cos(\theta - \phi)\sqrt{1 + \left(\frac{\sinh\theta}{\cosh\theta}\right)^2} \tag{8}$$

Appendix G:

Test Data

Table 5. Data for air-flow tests[a]

Blower speed (rpm)	Tip-clearance (in.)	Degree of throttling	Average torque (lb-ft)	Static pressure (in.H_2O)	Volume (cfm)	Power (hp)
500	1/32	0	8.01	1.14	341	0.76
	1/16		5.87	1.07	337	0.56
	1/8		4.59	1.05	333	0.42
	1/4		1.82	1.07	329	0.17
	3/8		2.97	1.06	324	0.28
	1/2		5.06	1.07	328	0.49
600	1/32	0	8.78	1.66	412	0.99
		1/2	5.60	1.52	326	0.59
		3/4	6.08	1.61	208	0.69
		Full	6.55	1.58	---	0.77
	1/16	0	5.94	1.50	404	0.67
		1/2	2.50	1.54	323	0.28
		3/4	6.08	1.59	200	0.67
		Full	7.22	1.57	---	0.76
	1/8	0	6.82	1.49	395	0.80
		1/2	7.22	1.51	335	0.78
		3/4	7.42	1.56	206	0.85
		Full	5.20	1.62	---	0.59
	1/4	0	3.64	1.49	386	0.42
		1/2	8.10	1.51	333	0.93
		3/4	7.22	1.51	200	0.81
		Full	4.25	1.59	---	0.49
	3/8	0	3.38	1.52	398	0.39
		1/2	4.72	1.55	321	0.54
		3/4	10.3	1.52	202	1.09
		Full	6.28	1.62	---	0.68
	1/2	0	7.56	1.48	393	0.86
		1/2	3.17	1.49	317	0.39
		3/4	7.42	1.57	193	0.91
		Full	3.58	1.52	---	0.41

[a]Static pressure, volume, and power are corrected for speed variation.

Table 5. (Continued)

Blower speed (rpm)	Tip-clearance (in.)	Degree of throttling	Average torque (lb-ft)	Static pressure (in.H_2O)	Volume (cfm)	Power (hp)
700	1/32	0	8.44	2.27	483	1.13
	1/16		4.93	2.09	472	0.68
	1/8		8.91	2.04	467	1.22
	1/4		4.18	2.07	460	0.57
	3/8		3.04	2.09	468	0.42
	1/2		8.78	2.03	458	1.18
800	1/32	0	6.41	2.95	541	0.98
		1/2	3.44	2.82	447	0.56
		3/4	4.93	2.81	275	0.76
		Full	5.06	2.92	---	0.78
	1/16	0	6.28	2.72	542	0.96
		1/2	2.36	2.83	446	0.38
		3/4	4.79	2.84	273	0.72
		Full	6.28	2.90	---	0.98
	1/8	0	10.1	2.69	538	1.57
		1/2	5.54	2.76	444	0.83
		3/4	6.62	2.87	279	1.00
		Full	5.74	2.89	---	0.84
	1/4	0	3.85	2.71	520	0.60
		1/2	8.44	2.72	446	1.29
		3/4	7.36	2.81	275	1.16
		Full	1.69	2.85	---	0.26
	3/8	0	2.16	2.73	533	0.33
		1/2	10.5	2.80	439	1.57
		3/4	10.6	2.79	276	1.54
		Full	6.28	2.92	---	0.98
	1/2	0	8.10	2.71	527	1.24
		1/2	1.82	2.69	438	0.29
		3/4	5.06	2.74	263	0.74
		Full	3.54	2.75	---	0.56

Table 5. (Continued)

Blower speed (rpm)	Tip-clearance (in.)	Degree of throttling	Average torque (lb-ft)	Static pressure (in.H_2O)	Volume (cfm)	Power (hp)
900	1/32	0	6.14	3.52	606	1.05
	1/16		6.55	3.44	605	1.12
	1/8		11.8	3.40	602	2.02
	1/4		11.9	3.50	592	2.14
	3/8		2.23	3.45	599	0.39
	1/2		8.30	3.46	599	1.42
960	1/32	0	8.37	4.02	644	1.54
		1/2	4.93	4.04	538	0.91
		3/4	4.18	4.06	332	0.77
		Full	6.14	4.20	---	1.13
	1/16	0	7.56	3.97	650	1.38
		1/2	4.05	4.03	537	0.74
		3/4	5.26	4.09	332	0.97
		Full	8.10	4.15	---	1.47
	1/8	0	12.6	3.87	645	2.30
		1/2	6.41	4.00	542	1.18
		3/4	6.62	4.03	330	1.20
		Full	6.75	4.12	---	1.22
	1/4	0	13.8	3.88	634	2.51
		1/2	11.3	4.03	544	2.11
		3/4	7.96	3.98	326	1.45
		Full	1.82	4.04	---	0.33
	3/8	0	2.23	3.93	639	0.41
		1/2	10.1	3.98	534	1.84
		3/4	10.8	4.08	333	1.98
		Full	7.76	4.14	---	1.41
	1/2	0	8.30	3.88	635	1.52
		1/2	3.71	3.84	522	0.68
		3/4	6.01	3.96	325	1.11
		Full	6.41	4.01	---	1.19

Table 6. Data for preliminary forage tests

Tip-clearance (in.)	Nominal blower speed (rpm)	Observed speed (rpm)	Average torque (lb-ft)	Feed rate (ton/hr)	Power (hp)	Hp-hr per ton
1/32	600	588	54.0	8.3	6.0	0.73
		588	58.0	9.9	6.5	0.66
		581	60.1	10.7	6.6	0.62
	750	728	61.4	10.7	8.5	0.80
		735	66.8	11.5	9.4	0.81
		714	64.8	12.3	8.8	0.72
		714	74.2	13.1	10.1	0.77
	850	824	89.1	13.1	14.0	1.06
		833	100.6	14.8	16.0	1.07
		820	110.6	18.1	17.3	0.96
		820	113.8	18.9	17.8	0.94
	950	931	125.4	18.1	22.2	1.23
		920	148.5	18.9	26.0	1.38
		915	165.0	19.8	28.8	1.45
		909	183.2	21.1	31.7	1.51
1/16	600	540	135.3	23.0	13.9	0.60
		521	148.5	24.7	14.7	0.60
		510	150.2	26.4	14.6	0.55
	950	862	188.1	31.6	30.9	0.98
		847	199.6	33.6	32.2	0.96
		820	206.2	35.5	32.2	0.91
1/8	600	530	138.6	20.6	14.0	0.68
		538	135.3	22.2	13.9	0.62
		556	165.0	24.7	17.5	0.71
	950	904	204.6	32.0	35.2	1.10
		904	222.8	33.6	38.4	1.14
		877	236.0	36.4	39.4	1.08

Table 6. (Continued)

Tip-clearance (in.)	Nominal blower speed (rpm)	Observed speed (rpm)	Average torque (lb-ft)	Feed rate (ton/hr)	Power (hp)	Hp-hr per ton
1/4	600	564	95.7	14.8	10.3	0.69
		556	107.2	16.5	11.4	0.69
		556	146.8	18.1	15.5	0.86
	950	898	145.2	26.4	24.8	0.94
		843	178.2	28.8	28.6	0.99
		852	229.4	32.1	37.2	1.16
3/8	600	556	102.3	22.2	10.8	0.49
		551	113.8	23.8	11.9	0.50
		528	125.4	26.4	12.6	0.48
	950	915	158.4	32.0	27.6	0.86
		882	204.6	38.3	34.4	0.90
		712	239.2	44.7	32.4	0.72
1/2	600	605	16.5	6.6	1.9	0.29
		600	19.8	7.0	2.3	0.32
	750	721	64.4	16.5	8.8	0.54
		714	71.0	18.1	9.6	0.53
		711	87.4	19.8	11.8	0.60
		704	108.9	21.1	14.6	0.69
	850	862	92.4	22.2	15.2	0.68
		811	108.9	23.8	16.8	0.70
		815	123.8	24.4	19.2	0.79
		829	148.5	25.4	23.4	0.92
	950	938	95.7	28.8	17.1	0.59
		926	130.4	29.6	23.0	0.78
		915	143.6	32.0	25.0	0.78

Table 7. Data for power-requirement tests with a feed rate of 20.6 tons per hour and nominal blower speed of 800 rpm[a]

Tip-clearance (in.)	Average torque (lb-ft)	Blower speed (rpm)	Power (hp)	
1/32	242.6	769	35.5	
	232.6	773	34.2	
	216.2	769	31.7	33.8[b]
1/16	183.2	769	26.8	
	257.4	765	37.5	
	226.0	777	33.4	32.6
1/8	201.3	781	29.9	
	204.6	777	30.3	
	193.0	769	28.3	29.5
1/4	165.0	777	24.4	
	155.3	789	23.3	
	203.0	773	29.9	25.9
3/8	132.0	798	20.0	
	114.0	781	16.9	
	139.6	785	20.9	19.3
1/2	100.8	789	15.1	
	113.5	789	17.1	
	95.0	794	14.4	15.5

[a]Tests were on alfalfa hay with 18 per cent moisture content and 3-inch length of cut.

[b]Underlined values are power averages.

Table 8. Data for power-requirement tests with a feed rate of 14.5 tons per hour and nominal blower speed of 800 rpm[a]

Tip-clearance (in.)	Average torque (lb-ft)	Blower speed (rpm)	Power (hp)	
1/16	133.8	781	19.9	
	131.7	768	19.2	
	138.8	789	20.8	20.0[b]
1/4	121.3	789	18.2	
	114.8	785	17.1	
	110.7	789	16.6	17.3
3/8	83.3	798	12.6	
	82.2	798	12.5	
	82.5	798	12.5	12.5
1/2	70.8	798	10.7	
	71.1	798	10.8	
	77.6	798	11.8	11.1

[a]Tests were on alfalfa hay with 23 per cent moisture content and 3-inch length of cut.

[b]Underlined values are power averages.

Table 9. Data for power-requirement tests with a feed rate of 14.5 tons per hour and nominal blower speed of 750 rpm[a]

Tip-clearance (in.)	Average torque (lb-ft)	Blower speed (rpm)	Power (hp)	
1/32	184.8	740	26.0	
	178.2	714	24.2	25.1[b]
1/16	137.6	714	18.7	
	181.5	723	24.9	
	160.0	723	22.0	21.9
1/8	143.9	723	19.8	
	147.3	714	20.0	19.9
1/4	155.4	723	21.4	
	139.6	723	19.2	
	130.0	714	17.7	19.4
3/8	108.9	740	15.3	
	90.1	740	12.7	
	129.4	740	18.2	15.4
1/2	81.8	750	11.7	
	76.7	750	10.9	
	78.9	750	11.3	11.3

[a]Tests were on alfalfa hay with 24 per cent moisture content and 3-inch length of cut.

[b]Underlined values are power averages.